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STUDY OF EFFECT OF A NON-NEWTONIAN OIL ON FRICTION AND ECCENTRICITY RATIO OF A PLAIN JOURNAL BEARING

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SUMMARY

The present report describes an experimental and analytical investigation of the hydrodynamic effects of lubrication on friction and load capacity of a plain journal bearing using a non-Newtonian oil in which the viscosity varies nonlinearly with shear rate as well as with temperature. Very high shear rates occur in a loaded bearing and for a non-Newtonian oil result in a marked reduction in viscosity and bearing friction. Evaluation of the effect on load capacity is difficult to obtain with sufficient accuracy either experimentally, because of the minute changes in eccentricity at high loads, or analytically, because the nonlinear viscosity affects fundamental concepts which require approximation. The experimental effects shown are of the same order of magnitude as the normal spread of the data and are not in agreement with the analytical work; however, when compared at high rotative speeds, the experimental data appear to indicate an advantage for the non-Newtonian oil either in friction or load capacity, depending on the viscosity chosen for comparison. This report is presented to bring out the complexities encountered in an attempt to understand the behavior of a non-Newtonian oil in a plain bearing and to indicate the need of confirmation from other sources before the experimental indication is accepted as true.

INTRODUCTION

Lubricating oils with certain additives may be classed as non-Newtonian when their viscosity varies with shear rate as well as with temperature, in contrast with petroleum oils which are normally Newtonian in that the viscosity is independent of shear rate at a given temperature. As shown in figures 1(a) and 1(b), a non-Newtonian fluid designated as ASTM 104 shows a reduction of viscosity of the order of 40 percent at high shear rates.

The use of a non-Newtonian oil in which the viscosity varies nonlinearly with shear rate introduces many fundamental complexities in hydrodynamic performance. The shear rate in a journal bearing varies from point to point in the oil film and is a vector sum of a component due to rotational velocity divided by film thickness plus a component due to the slope of a curved velocity profile which depends on the pressure gradient in the streamline direction. With a Newtonian oil the viscosity is independent of these variables and is assumed constant at a given temperature, but with a non-Newtonian oil the viscosity becomes variable from point to point in the oil film. Figure 1 gives the local viscosity if the local shear rate is known; but to obtain an equivalent viscosity suitable for predicting performance would require a corresponding shear rate which is unknown. In this report, the equivalent viscosity of the non-Newtonian oil should be regarded as an unknown to be determined.

With a non-Newtonian fluid the usual assumption of a parabolic velocity distribution in a pressure-induced flow requires modification because the viscosity varies with the local slope or shear rate of the velocity profile. The forces which establish the parabolic profile are somewhat altered, and the parabolic shape is modified. This effect is reflected in methods of correlating non-Newtonian viscosity data from capillary viscometers which have a modified parabolic profile with that from rotating-shear-type viscometers which have a normally linear velocity profile.

Both types of velocity profile occur simultaneously in a journal bearing. The modified parabolic profile is primarily related to the film pressure and the load capacity, while the linear profile is primarily related to the bearing friction portion of the power loss, the remainder of the power loss being a couple related to the eccentricity ratio and the total load.

Tests of non-Newtonian oils in journal bearings are needed for comparison with viscosity data from high-shear-rate viscometers of both the linear and capillary type. This requires measurement of eccentricity ratio as well as of friction. An experimental and analytical investigation of the effect of a non-Newtonian oil on the friction and eccentricity ratio of a plain journal bearing was conducted at Cornell University under the sponsorship and with the financial assistance of the National Advisory Committee for Aeronautics. The early experimental results were in the form of curves based on the viscosity at low shear rate. Later an analytical treatment was attempted as the most fruitful avenue of approach. This led to the concept of analytically equivalent viscosities differing for friction and for load. These were later supplemented by similar equivalent experimental viscosity values which showed possible speed effects.

In the investigation, measurements of eccentricity and friction in a journal bearing were made for a wide range of loads, speeds, temperatures, and shearing rates using two non-Newtonian oils as well as a normal oil. An analytical investigation of hydrodynamic effects was also made in an attempt to increase understanding of the effect of non-Newtonian lubricating oils in journal bearings. Since the experimental data deal with an effect but not the cause, the description of the analytical investigation is useful in revealing fundamental considerations in more detail. The nonlinear variation of viscosity with shear rate causes fundamental difficulties which require approximations and numerical integration in order to obtain equivalent viscosities analytically for use in estimating friction and load.

In order to permit direct comparison of the experimental data, dimensionless parameters are used which include an assumed viscosity for the non-Newtonian ASTM 104 oil. By assuming different viscosities either the friction or the load capacity may be brought into agreement with similar data for the normal ASTM 101 oil. Close agreement indicates that the assumed viscosity approximates an equivalent viscosity intended for estimating performance.

The sources of error in both the experimental and analytical techniques are also discussed.

SYMBOLS

C_n	capacity number, $C_n = S\left(\frac{l}{d}\right)^2 = \frac{\mu N'}{p}\left(\frac{d}{c_d}\right)^2\left(\frac{l}{d}\right)^2 = \frac{1}{N_L}$
C_{n_o}	capacity number based on μ_o at low shear rate
c_d	diametral clearance, $2c_r$, in.
c_d/d	diametral clearance ratio
c_r	radial clearance, $c_r = c_d/2$, in.
d	bearing diameter, in.
e	eccentricity of journal and bearing axes, in.
F	circumferential bearing friction force under load, lb

F/F_o	friction number
F_o	circumferential bearing friction force at zero load, $2\pi^2 \mu N' l d \left(d/c_d\right)$, lb
h	local fluid film thickness, in.
l	bearing length, in.
L	subscript denoting load
l/d	length-diameter ratio
N'	journal speed, rps
N_L	load number, $N_L = \frac{1}{C_n} = \frac{p}{\mu N'}\left(\frac{c_d}{d}\right)^2\left(\frac{d}{l}\right)^2$
n	eccentricity ratio, e/c_r
P	applied central load, lb
p	unit pressure on projected area, lb/sq in.
R_θ, R_z	shear rate in rotational and axial directions, respectively, sec^{-1}
$R_{\theta_{av}}$	average rotational shear rate, sec^{-1}
S	Sommerfeld number, $\frac{\mu N'}{p}\left(\frac{d}{c_d}\right)^2$
T	temperature, ^{o}F
U	surface speed of journal, in./sec
z	axial coordinate
θ	angular location measured from maximum film thickness (fig. 2), deg
θ_e	circumferential location around bearing where local viscosity of non-Newtonian fluid is equal to equivalent viscosity μ_{e_f} of Newtonian fluid

θ_{e_f}, θ_{e_L}	angular location for determining analytical average shear rate for friction and load, respectively, deg
μ	absolute viscosity of fluid (centipoises/6.9×10^6) (variable with shear rate for non-Newtonian fluids), reyns
μ_o	absolute viscosity of fluid determined by low-shear-rate viscometers, reyns
μ_{e_f}, μ_{e_L}	equivalent viscosities for friction and load, respectively, determined analytically
μ_{exp_f}, μ_{exp_L}	equivalent viscosity for friction and load, respectively, determined experimentally
$\bar{\mu}$	constant viscosity
τ	shear stress

ANALYSIS

The mathematical analysis of the hydrodynamic performance of a journal bearing with a non-Newtonian fluid was undertaken with a two-fold purpose:

(1) To determine mathematically whether the reduced viscosity due to shearing rate has a different effect on reducing bearing load capacity than it has on reducing the bearing friction force and to evaluate this difference for comparison with the difference which seems to be apparent from the experimental data

(2) To provide a means for determining a representative value of viscosity for the individual experimental runs in order that certain experimental load numbers and friction numbers may be evaluated using an analytical value of viscosity which is more realistic than the low-shearing-rate viscosity

Viscosity Data

The viscosity variation with shearing rate and temperature used in the analysis is shown in figure 1. The viscosity data in figure 1(a) were obtained (refs. 1 and 2) in high-shear-rate capillary viscometers at 100°, 150°, and 210° F and were cross-plotted by interpolation in figure 1(b). Figure 1(a) shows temporary losses in viscosity at high shear rates. No permanent loss in viscosity was detected by analysis in the same viscometers of samples taken after the journal bearing tests were completed.

In determining the viscosity in high-shear-rate capillary viscometers, Fenske, Klaus, and Dannenbrink in reference 1 mention the probable modification of the parabolic velocity profile for a non-Newtonian oil and originally plotted the data against the maximum shear rate at the capillary tube wall based on the Hagen-Poisenille parabolic relationship which is correct for Newtonian oil. In reference 2, they compared their data from the capillary viscometer with the data obtained by Needs (ref. 3) on the same oil in a tapered-plug viscometer.

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Needs' apparatus consisted of concentric cylinders, one of which rotated to shear the fluid in a linear velocity profile, and the shearing rate was determined from the surface velocity of the rotating cylinder and the film thickness. In the tapered-plug viscometer the entire sample of fluid is at the same shearing rate, whereas in the capillary viscometer the sample has zero shearing rate at the tube center and a high shearing rate at the wall. By comparing the data from both viscometers, Fenske, Klaus, and Dannenbrink show that the data correlate well if for the capillary data half of the maximum shearing rate value is used rather than the total maximum value. The half value of the maximum shearing rate called the average shearing rate is used in figure 1(a). Thus, while the viscosity data in figure 1(a) were obtained in high-shear-rate capillary viscometers, they correlate well with the tapered-plug data which appear to be true values of viscosity and shearing rate.

The problem of correlation of high-shear-rate viscosity data from different types of viscometers is closely related to the present problem since in a loaded bearing the circumferential velocity profiles are more nearly like those in the tapered-plug viscometer, and those in a capillary viscometer are the closest available to the axial velocity profiles in a journal bearing.

Method of Attack

The following logic based on hydrodynamic theory is used to compare mathematically the frictional and load-carrying capacity of a bearing having a non-Newtonian fluid with that of one having a Newtonian fluid.

First, at some assumed eccentricity ratio n an integration is made for the total friction force F of the non-Newtonian oil including the variable viscosity based on variable shearing rates in the film. A given oil at a given temperature T and a given journal speed N' are assumed so that local viscosity may be determined from a curve in figure 1(a).

Second, an integration for F is made assuming that viscosity is constant as in a Newtonian fluid but has an unknown value of equivalent viscosity for friction μ_{e_f}. Eccentricity ratio, journal speed, and the bearing dimensions including clearance are the same as for the non-Newtonian case. By equating the two expressions for F the equivalent viscosity μ_{e_f} is determined for the Newtonian fluid which gives the same total friction as for the non-Newtonian fluid.

The same procedure is employed in the integrations for total bearing load P to determine the equivalent viscosity for load μ_{e_L} which gives the Newtonian fluid the same load-carrying capacity as that of the non-Newtonian fluid. In the non-Newtonian case the same oil at the same temperature and the same eccentricity ratio are used as in the friction determination, except that viscosity is determined by the shear rate in the axial direction.

Thus, two values of equivalent viscosity are obtained, μ_{e_f} for equal friction force and μ_{e_L} for equal bearing load. A comparison of the two values of μ_e leads to a conclusion as to the relative advantage or disadvantage of using a non-Newtonian fluid instead of a normal one.

If μ_{e_L} for equal loads is greater than μ_{e_f} for equal frictions, then the non-Newtonian fluid analytically has the greater load-carrying capacity when the bearing friction forces of the two fluids are equal. Since for a Newtonian fluid the viscosity is the same for determining either F or P, the load capacity of the Newtonian fluid is dependent on μ_{e_f} and must be raised to μ_{e_L} to equal the load capacity of the non-Newtonian fluid.

Evaluation and Comparison of Viscosities

Evaluation of equivalent viscosity for friction.- The integral equation used to determine the friction force F of the bearing with the non-Newtonian fluid is that from the short-bearing approximation (ref. 4). In this solution, the velocity profiles are assumed to be of the forms shown in figure 2 in which the circumferential profiles are linear and the axial profiles are parabolic.

Referring to figure 2, it may be seen that for a loaded bearing at some eccentricity e (eccentricity ratio n), the shearing rate in the circumferential direction is variable so that the viscosity is also variable if the fluid is non-Newtonian. The shearing stress in the fluid depends on both the local shearing rate and local viscosity. An integration around the bearing with respect to θ gives the friction force F of the bearing, F being a circumferential force vector. In place of an integration, a numerical summation is required since the viscosity is a complex function of shearing rate. Assuming a fixed value of eccentricity, a fixed journal speed, and fixed bearing dimensions, the friction force F may be determined for the non-Newtonian oil at one of the temperatures in figure 1(a).

The same integral expression for friction force may be used for a normal oil except that the viscosity is constant and not dependent on shearing rate. If the same eccentricity, journal speed, and bearing dimensions are used in the integral expression as are used for the non-Newtonian oil, then the friction force is determined except for its dependency on viscosity. However, by equating the expressions for F of the two oils, the equivalent viscosity μ_{e_f} of the normal oil may be obtained which gives the same friction force for the two oils.

In figure 3 is shown a comparison of the shearing-stress distributions in a non-Newtonian oil film and in an equivalent Newtonian oil film when the friction forces F are equal. Even though the fixed eccentricity is high, it may be seen that shearing-stress distributions are not greatly different.

In table 1 are shown values of μ_{e_f} obtained by calculations for various values of fixed journal speed, oil temperatures, and eccentricity ratio. In all cases, the bearing dimensions such as length, diameter, and clearance are the same. The speeds of 1,000 and 8,000 rpm, and the oil temperatures of 131° and 169° F are typical values at the extremes of the experimental runs. As would be expected because of the effect of increased speed on increasing shearing rate, μ_{e_f} decreases as the journal speed is increased. Increasing the eccentricity ratio also decreases μ_{e_f}.

Referring to figure 3, θ_e is the circumferential location around the bearing where the local viscosity of the non-Newtonian fluid is equal to the equivalent viscosity μ_{e_f} of the equivalent Newtonian fluid. Thus, the shearing rate at this location is the representative shearing rate of the film at which the equivalent viscosity may be determined from figure 1(a). In table 1 are shown the calculated values of θ_e corresponding to the conditions at which μ_{e_f} values were calculated. In figure 4 are shown plots of θ_e against eccentricity ratio n from the table.

Figure 4 is interesting in that the θ_{e_f} data for the extreme speeds of 1,000 and 8,000 rpm with correspondingly different temperatures fall on a common line. This curve has the advantage that it may be used to determine the equivalent viscosity μ_{e_f} at any speed and oil temperature in the range of the experimental runs by a simple method. Using the eccentricity ratio n in the experiment, θ_{e_f} may be determined from the curve and the representative shearing rate may be determined at θ_{e_f}; then, from this shearing rate and the experimental oil temperature, μ_{e_f} may be determined from figure 1(a).

Evaluation of equivalent viscosity for bearing load.- The determination of the load P which a non-Newtonian film may support is relatively more complex than the determination of the friction force F. Although the load P is determined from an integration of local film pressure, a prior integration is required since the variable shearing rate and viscosity influence the pressure distribution.

According to the short-bearing approximation, the solution of the differential equation for pressure yields the pressure distribution based on the pressure flow in the axial or z-direction. In figure 2 are shown the parabolic velocity profiles in the axial direction and the corresponding parabolic axial film pressure distribution for a normal fluid. For the non-Newtonian fluid, a parabolic velocity profile is also assumed. For a parabolic velocity profile, the shearing stress and shearing rate are linear so that the average shearing rate is one-half the maximum shearing rate at the wall. The average shearing rate thus corresponds with that of the capillary viscometer data of figure 1(b).

The average shearing rate in the axial direction varies from zero at the bearing center to a high value at the bearing ends so that there is a correspondingly great variation in viscosity from bearing center to the ends. However, in determining the axial pressure distribution the product μz is used to determine the pressure gradient dp/dz. As shown in figure 5(a), the product μz is only slightly curvilinear even though the variation in μ is great. The straight dashed line shown is an approximation to the slightly curved line and represents the μz distribution with constant viscosity $\bar{\mu}$. The straight line is arbitrarily drawn through $z = 0.8(l/2)$ so that the areas under the two curves are equal. The axial pressure distribution which results is parabolic as shown in figure 5(b).

Thus, the axial pressure distribution is determined by evaluating the average shearing rate at $z = 0.8(l/2)$ and determining $\bar{\mu}$ from the appropriate viscosity curve in figure 1(a). However, at each circumferential station θ the shearing rate and $\bar{\mu}$ are different because of the eccentricity of the bearing. To determine the load P, an integration of local pressure circumferentially is required. Such an integration must be made numerically because of the complex relation of $\bar{\mu}$ with shear rate.

The integrations for determining the load P for a normal fluid are relatively simpler because viscosity is a constant. By equating the load P of the normal fluid to that of the non-Newtonian fluid at fixed conditions of speed, eccentricity ratio, and bearing dimensions, the equivalent viscosity μ_{e_L} may be determined of an equivalent Newtonian fluid which gives the same load capacity as the non-Newtonian fluid.

In table 2 are given the calculated values of μ_{e_L} for fixed values of journal speed, oil temperature, and eccentricity ratio. These may be compared with the values of μ_{e_f} in table 1 for the same fixed conditions. In table 2 are also shown the calculated values of the circumferential location θ_e at which the non-Newtonian viscosity $\bar{\mu}$ is equal to the equivalent viscosity μ_{e_L} of the equivalent normal fluid when both fluids support the same load. In figure 6 is shown the plot of θ_e against eccentricity ratio, and, as shown, the data for the extreme speeds of 1,000 and 7,000 rpm fall on a common line.

The curve of figure 6 serves the same role in determining μ_{e_L} as the curve of figure 4 serves in determining μ_{e_f}. If eccentricity ratio is known, θ_{e_L} may be determined from the curve and the shearing rate may be calculated at $z = 0.8(l/2)$ and $\theta = \theta_{e_L}$; then μ_{e_L} may be determined from the calculated shearing rate using the appropriate viscosity curve in figure 1(a).

Comparisons of μ_{e_f} and μ_{e_L}.- In table 3 the calculated values of equivalent viscosities μ_{e_f} and μ_{e_L} are compared for the same eccentricity ratios, the same journal speeds, and the same non-Newtonian oil at the same temperature. As shown, the values of μ_{e_L} are greater than the μ_{e_f} values at low eccentricity ratios, but at the high eccentricity ratios the reverse is indicated.

Referring to table 3 and the data shown for a journal speed of 1,000 rpm and an eccentricity ratio of $n = 0.3$, the equivalent viscosity is $\mu_{e_f} = 2.05 \times 10^{-6}$ reyn. A normal oil could be chosen which would have the same viscosity. For both oils in the same bearing at the same conditions, the friction torque should be the same analytically. The load which the normal oil would support would be proportional to the viscosity $\mu = 2.05 \times 10^{-6}$ reyn, but the load supported by the non-Newtonian oil would be proportional to that of a normal oil having the higher viscosity of $\mu = \mu_{e_L} = 2.32 \times 10^{-6}$ reyn. Thus, the non-Newtonian oil has the greater load capacity when μ_{e_L} is greater than μ_{e_f}.

As shown in table 3, at $n = 0.3$ the non-Newtonian oil has a 13 percent greater load capacity than does the Newtonian oil which gives the same friction torque. At $n = 0.9$, however, the load capacity of the non-Newtonian oil is 12 percent less. Table 4 shows a comparison of μ_{e_f} and μ_{e_L} with μ_o at low shear rate and the asymptotic values at high shear rates.

Conclusions Drawn From Analysis

The conclusion to be drawn from the calculated data in table 3 is that the non-Newtonian fluid may or may not be superior to a Newtonian one depending on the eccentricity ratio at which the bearing is to operate. For moderately loaded bearings in which the eccentricity ratio is approximately 0.6, there is relatively little advantage in using one of these fluids instead of the other.

It is difficult to accept the results of the analytical study because of the lack of confirmation from the experimental data. As discussed in another section of this report, the experimental data show a greater load capacity for the non-Newtonian oil for all values of eccentricity ratio. Moreover, the experimental data also show that the load-capacity increase of the non-Newtonian fluid over that of the Newtonian one becomes greater as the eccentricity is increased, a trend opposite to that predicted in table 3.

EXPERIMENTS

Friction and eccentricity experiments were made using both Newtonian and non-Newtonian oils in the same bearing at similar conditions in order to give comparable data. The test bearing, which had a $1\frac{3}{8}$-inch diameter and a $1\frac{3}{8}$-inch length or $l/d = 1.0$ and a 0.00355-inch diametral clearance, was located centrally between two support bearings in the test machine. Load was applied to the test bearing by a hydraulic pressure capsule. Journal speeds were from 1,000 to 8,000 rpm. The oils, supplied by the Petroleum Refining Laboratory of the Pennsylvania State University, were a normal oil ASTM 101 (PRL 3375) and two non-Newtonian oils ASTM 103 and ASTM 104 (PRL 3373 and PRL 3374). Viscosity measurements of the oils were made at Pennsylvania State University in a high-shear capillary viscometer as reported by Fenske, Klaus, and Dannenbrink in references 1 and 2. Data for the non-Newtonian ASTM 103 oil are not presented in this paper since this oil had less effect on a bearing than did the non-Newtonian ASTM 104 oil.

The non-Newtonian ASTM 104 oil consisted of a low-viscosity mineral-oil-base blend thickened by adding 5 percent of a high-molecular-weight polymethacrylate ester and had the same viscosity at 100° F as did the straight Newtonian mineral oil (ASTM 101) used for comparison when both were measured in a low-shear-rate viscometer. This viscosity at low shear rate approximates the intercept of the viscosity line with the zero-shear-rate line and is called "zero-shear-rate" viscosity μ_0. A comparison of the viscosity of the two oils with temperature at a low and a high shear rate (250,000 seconds^{-1}) is shown in figure 1(c).

The principal measurements made were friction and eccentricity ratio. The friction of the bearing was measured by a hydraulic torquemeter. Measurement of eccentricity was made with a system of levers and 0.0001-inch dial indicators, two at each end of the test bearing giving orthogonal components of the eccentricity. The system of levers measured the position of the journal relative to the bearing by rubbing contact on the journal surface. Average bearing temperature was measured from several thermocouples imbedded in the bearing wall near the film. The apparatus and test methods were the same as those used in the tests reported in reference 5 in which figures and a full description of the test procedure are given.

Test Procedure

The tests were run at constant journal speeds with increasing loads because this procedure results in a minimum change of bearing temperature which in turn has a minimum temperature effect on eccentricity measurements. Friction experiments were made separately from the eccentricity experiments to eliminate the friction of the rubbing surfaces of the eccentricity measuring system. Both friction and eccentricity tests were made in each direction of journal rotation, and the data were averaged and corrected for estimated shaft bending and clearance changes with temperature.

The tests were begun with an accurately bored bearing with a diametral clearance of 0.0018 inch which had been used in previous tests with SAE 10 oil. The initial tests in this investigation with the ASTM 101 oil were nearing completion at high load and 10,000 rpm when a seizure, attributed to insufficient clearance, damaged the bearing slightly. The damaged bearing was refinished by honing to a clearance of 0.00355 inch with some loss of accuracy of the bore. A complete series of tests was then made with the enlarged clearance to obtain data giving a direct comparison of the oils in the same bearing under similar conditions.

The test machine and piping systems were thoroughly flushed with Varsol, blown out, and drained in partial disassembly before each change of the test oils to avoid contamination or dilution. Samples of the test oils were taken at the end of the tests and forwarded to the Petroleum Refining Laboratory for analysis. A report from that laboratory on the ASTM 101 and ASTM 104 oils indicated that ". . . there is no significant evidence from the properties of the used samples, of oxidation deterioration, permanent viscosity decrease, or evaporation."

Variables.- The dimensionless parameters are those determined from the short-bearing approximation (ref. 4) and are in a form adopted in reference 5. In figure 7, $n = e/c_r$ is the eccentricity ratio determined from the measured eccentricity e and the measured radial clearance c_r. The load number includes the other measured bearing variables where p is the unit load on the bearing, N' is the journal speed, μ is a particular value of oil viscosity as indicated by subscripts, and d, l, and c_d are the diameter, length, and diametral clearance of the bearing, respectively. The friction number F/F_o is the ratio of the measured friction force F for a loaded bearing to the calculated Petroff friction force F_o of the bearing without load and F_o includes the bearing dimensions, journal speed, and value of fluid viscosity indicated in the figure.

The oil viscosity at low shear rate at the averaged temperature in the oil film is μ_o which is the correct viscosity for the Newtonian oil but neglects the viscosity reduction with higher shear rate for the non-Newtonian oil. A value of viscosity μ_{exp_f} is determined from bearing friction at high shear rates. The values of μ_{e_f} and μ_{e_L} are analytically estimated at high shear rate from curves derived in the "Analysis" section.

The temperature variation of the viscosities of both oils at low shear rate was obtained from a logarithmic chart in which a straight line joins the points given in table 5, which lists the data taken from the tags of the containers of the test oils giving viscosity, density, ASTM chart slope, and viscosity index.

Experimental Data

Figure 1 illustrates the viscosity changes both with temperature, obtained from table 5, and with shear rate for the non-Newtonian oil, obtained from references 1, 2, and 3 as described in the "Analysis" section.

Figures 7 to 10 show the experimental data for friction and eccentricity ratio of the two oils. The data are shown on the basis of assumed viscosities for the non-Newtonian oil, since the average shear rate and the average viscosity are unknown, in order to show the accuracy of the assumed viscosity value.

The dimensionless parameters used in the graphs should largely correct for all operating conditions for the Newtonian oil so that all data points for the Newtonian oil should fall near a single line, except for the normal spread of the data. Thus the differences between the curves, neglecting the spread of the data, should be those due to the changes of the viscosity of the non-Newtonian oil.

<u>Curves based on μ_o</u>.- In figure 7(a), the experimental data for the ASTM 104 oil are plotted using the viscosity at low shear rate μ_o at the bearing temperature. Thus, the use of μ_o in the friction number and in the load number leads to fictitious values of these numbers for the 104 oil because the true viscosity is lowered by shearing action. Nevertheless, the curves in the figure show that at the speeds of 1,000, 2,500, 5,000, and 8,000 rpm the measured friction F is less for the non-Newtonian oil than it was for the Newtonian oil at the same temperature, presumably because the true viscosity is less than the viscosity at low shear rate. At the same time, the curves of eccentricity ratio at the speeds of 1,000, 2,500, 5,000, and 7,000 rpm (fig. 7(b)) show that the eccentricity ratio n differs only slightly between the two oils. The eccentricity curves appear to indicate that, neglecting the normal spread of the data, the average viscosity for estimating load μ_{exp_L}, in general, is fairly close to μ_o, but that the average viscosity for friction is considerably different.

<u>Curves based on μ_{exp_f}</u>.- Figure 8 was obtained graphically from figure 7 in order to alter the value of μ_o for the non-Newtonian oil to a value μ_{exp_f} which is more representative of the experimental friction data. The viscosity of the non-Newtonian oil when reduced by shear rate varies from point to point in the oil film, and an average viscosity is to be inferred from the experimental friction data. As shown in the upper graph in figure 11, the viscosity μ_o appears in the denominator of each parameter; and a change in viscosity moves a point along each parameter proportionally to produce a motion along a line through the origin. If the correct average viscosity were used for each point, the curve for the non-Newtonian oil should coincide with that for the Newtonian oil ASTM 101, so that the reduction in viscosity due to shear rate is related to the ratio AB/OB at each point.

In order to base the non-Newtonian friction data on a viscosity μ_{exp_f}, the change in the friction curve in figure 8 was obtained graphically by moving the points on the non-Newtonian friction curve along radial lines from A on the 104 curve to B on the 101 curve. The corresponding changes in the friction number and the load number are the vertical and horizontal coordinates of AB. In this way the friction curves for the two oils are made to coincide, and the eccentricity curves are altered only by the horizontal coordinate which changes the load number as shown in the lower curve in figure 11.

The resulting curves in figure 8 transfer the difference between the oils to a change in load capacity and may be interpreted as the difference in load capacity that would occur if a Newtonian oil were chosen which would give the same friction as the non-Newtonian oil.

<u>Curves based on μ_{e_f} and μ_{e_L}</u>.- In figure 9 are shown plots of the experimental data in which the viscosities in the friction and load numbers are calculated equivalent viscosities μ_{e_f} and μ_{e_L}. These are derived in the "Analysis" section of this report, where it is shown on the basis of certain assumptions that the equivalent viscosity μ_{e_f} related to the bearing friction is dependent on the shear rates in the circumferential direction of film flow, whereas μ_{e_L} related to load capacity depends primarily on the axial shear rates in the film.

The method of choosing μ_{e_f} and μ_{e_L} from figure 1(a) for the ASTM 104 oil stems from the analytical study presented in this report. From figures 4 and 6, the angular circumferential position θ_e is selected at which the representative shear rate may be calculated to determine the equivalent viscosity from figure 1(a). Figure 4 is used in determining μ_{e_f} and figure 6, in determining μ_{e_L}. Referring to figure 6 as an example, the eccentricity ratio is known from experimental measurement so that θ_{e_L} may be determined. The representative shear rate R_z for flow in the axial direction may then be calculated from the following equation:

$$R_z = \frac{6\pi N'}{c_d} \frac{n \sin \theta_{e_L}}{\left(1 + n \cos \theta_{e_L}\right)} 0.8 \frac{l}{2} \tag{1}$$

All factors in the equation are known from the experimental data. Using this calculated value of shear rate and the temperature measured in the experiments, the equivalent viscosity μ_{e_L} may be determined for the ASTM 104 oil from figure 1(a). The determination of μ_{e_f} is made in a similar manner except that figure 4 is used to determine θ_{e_f} and the representative shear rate R_θ for flow in the circumferential direction of flow is given by the following equation:

$$R_\theta = \frac{2\pi N'd}{c_d} \frac{1}{\left(1 + n \cos \theta_{e_f}\right)} \tag{2}$$

It may be seen that the curves for the friction runs for the ASTM 104 and ASTM 101 oils fall nearer together in figure 9 than they do in figure 7 where low-shear-rate viscosity is used. However, the lines for the two oils are not quite coincident, which indicates that the analytical method for determining μ_{e_f} and μ_{e_L}, while not exact, is in the correct direction for friction.

With regard to the curves for eccentricity ratio in figure 9, it would appear that these data should fall on a common line by using μ_{e_L}. However, these curves were nearly coincident on the basis of low-shear-rate viscosity μ_o as in figure 7, and it is obvious that they would separate horizontally by using a reduced viscosity μ_{e_L} for the ASTM 104 oil.

The experimental data in figures 8 and 9 show the same advantage for the non-Newtonian oil as they do in figure 7. The effect of introducing more representative values of viscosity has been merely to shift the curves. In figures 8 and 9 the curves show that at nearly the same friction force for the two oils, the non-Newtonian oil has the greater load capacity. In figure 7 they show that for the same load the non-Newtonian oil has the lesser friction.

Speed effect.- Figure 10 was obtained from figure 8, where the friction curves are equal, by cross-plotting the differences in eccentricity ratio for the different speeds. Apparently, a similar effect could be shown in friction on a basis of equal eccentricity ratio. Figure 10 serves to indicate that there is some basis for a belief that the effect of the non-Newtonian oil increases with speed, being negligible at 1,000 rpm and considerable at 7,000 rpm.

Viscosity loss.- Figure 12 compares the loss in the average viscosity indicated by the friction experiments with the high shear rate viscometer data in figure 1(a). Referring to figure 11, if μ_o is the viscosity at low shear rate, and μ is the correct viscosity of the non-Newtonian oil at a data point, making it fall on the ASTM 101 line, then the percentage change in viscosity is

$$\frac{\mu_o - \mu}{\mu_o} \times 100$$

If the load number is represented by $1/C_n$, then C_n is proportional to μ, and the percentage change in viscosity is

$$\frac{C_{n_o} - C_n}{C_{n_o}} \times 100$$

Multiplying both numerator and denominator by $1/\left(C_{n_o}C_n\right)$ gives the percentage change in viscosity

$$\frac{\left(1/C_n\right) - \left(1/C_{n_o}\right)}{1/C_n} \times 100$$

In figure 11 the abscissa of point A is $1/C_{n_o}$ and that of B is $1/C_n$ obtained by using the correct viscosity μ. Thus, the viscosity loss may be calculated from the load number of points A and B for each data point.

The loss in viscosity determined in this way represents the average viscosity loss for friction, since the actual viscosity reduction at any point depends on the local shear rates which are variable throughout the oil film in a loaded bearing.

In figure 12 the average viscosity loss for friction is compared with the loss obtained from the viscometer data in figure 1(a) by using an average shear rate in the rotational plane obtained analytically. A linear velocity profile is assumed so the shear rate is given by U/h, and the integral average used is

$$R_{\theta_{av}} = \frac{1}{2\pi} \int_0^{2\pi} \frac{U}{h} \, d\theta$$

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Substituting $h = c_r(1 + n \cos \theta)$ from reference 4 gives

$$R_{\theta_{av}} = \frac{U}{c_r} \frac{1}{\left(1 - n^2\right)^{1/2}} = \frac{2\pi N'd}{c_d} \frac{1}{\left(1 - n^2\right)^{1/2}} \tag{3}$$

The percent loss in the viscometer data in figure 1(a) varies only a little with temperature, and a solid line is drawn through points indicating the average loss at various shear rates.

The correlation of the average viscosity loss for bearing friction with the viscometer data using an average rotational shear rate corrected for the effect of the measured eccentricity n is reasonable with the largest scatter at 1,000 rpm.

RESULTS

The reduction in friction with increasing shear rate obtained experimentally gives a reasonable correlation with that predicted analytically as shown in figure 12. This supports the concept that the bearing friction in an eccentric bearing is primarily related to the linear velocity profile but requires correction for the effect of the eccentricity ratio on the average shear rate as shown.

The reduction in load capacity with increasing shear rate which seems to be indicated experimentally is much less than the analytical result, although neither method is of the accuracy required to warrant a definite conclusion.

The two methods disagree not only in the effect of speed but also in the effect of eccentricity ratio. Mathematical analysis indicates the load capacity of the non-Newtonian fluid to be increased at low eccentricities and decreased at high eccentricities, a reversing effect. However, the experimental evidence indicates a possibility of an opposite trend in which the increased load capacity of the non-Newtonian fluid, based on equal friction, becomes greater as the eccentricity ratio is increased. This is vividly demonstrated in the curves of figures 7 and 8. It would appear from the experimental data that, insofar as load capacity is concerned, the non-Newtonian fluid tends to maintain its low-shear-rate value, but, insofar as friction force is concerned, the viscosity is greatly reduced by the shearing effect.

Because of the lack of mutual confirmation between the experimental and analytical results, no definite conclusions should be drawn from either technique. Experimental techniques used for the measurement of eccentricity ratio are always open to question. The assumptions and approximations used in predicting analytically the hydrodynamic performance of a complex shear-rate-sensitive fluid in a journal bearing are equally questionable. The sources of error in each technique are treated in the following section.

DISCUSSION

The viscosities for friction and load obtained analytically show less change than, and have a different trend from, those obtained experimentally. Therefore, the analysis is valuable in revealing the problem in detail but does not provide an explanation for the experimental indication. There is the possibility that the experimental difference may be due to some viscous effect other than that introduced into the analysis which was based on the viscosity—shear-rate data in figure 1. The differences in eccentricity ratio at high loads are so minute for large changes of load that a considerable spread of experimental eccentricity ratio data is normally expected even for a single Newtonian oil. Comparison of two oils, one having a considerably different non-Newtonian viscosity, enhances the possibility of error in comparing differences in these data. Since the differences in question are of the same order of magnitude as the normal spread of the data, the experimental results should not be considered conclusive; however, the experimental differences appear to be regular rather than random and appear to indicate a viscosity for friction different from that for load at higher journal speeds. Since this can be interpreted to mean either a reduction in friction for the same load or an increase in load capacity for the same friction, there is a possibility of an advantage for the non-Newtonian oil although the experimental data are inconclusive. Experimentally the results indicate a possibility, rather than a proof, which should be more thoroughly investigated.

In view of the conflicting evidence, it is enlightening to review in detail the possible sources of difficulties in the experimental techniques employed and in the physical principles and assumptions used in the analysis. In the following paragraphs, the difficulties are discussed by listing arguments in support of the conclusion of the analysis on one hand and the arguments supporting the experimentally observed behavior on the other.

Arguments Supporting the Analysis

The arguments supporting the conclusions of the analysis are as follows:

(1) The analysis reveals little evidence at high shear rates to indicate that the effective load-carrying viscosity will be greatly different from the effective friction viscosity. At very high rotative speed nearly all of the fluid in the film will be at such a high shear rate that the viscosity will approach the reduced asymptotic value of the curves of figure 1(a), so that the viscosity should be effectively constant throughout the film and the behavior of the film should therefore be nearly Newtonian. At the very high journal speed, μ_{e_f} and μ_{e_L} should be of nearly the same value and both friction force and load capacity should be equally reduced. At the lower journal speeds, the values of μ_{e_f} and μ_{e_L} should be different because of the influence of the curvilinear portion of the viscosity curves in figure 1(a). The mathematical analysis is in support of this trend, but the experimental data are not since they show a great difference in μ_{e_f} and μ_{e_L} at the high journal speeds.

(2) Most investigators have found experimental measurements of eccentricity difficult for thin films, with specific differences of uncertain accuracy. In figure 8, for example, the differences in eccentricity ratio are due to differences in eccentricity of about 0.0001 inch, a small value to measure accurately considering the rotation, elastic deflection, thermal expansion, etc. In past experiments, although plots of measured eccentricity against load number are smooth curves, repeated runs at the same conditions have shown some scatter of the data.

(3) In earlier experiments, curves of measured eccentricity against load number have varied somewhat for large changes in viscosity, speed, and bearing clearance even though theoretically this is accounted for in the dimensionless parameters. Although the variations of the eccentricity curves have not been great, they are great enough to question whether the accuracy is adequate for measuring the differences between the eccentricities of the non-Newtonian and Newtonian oils.

(4) In the early portion of the experimental program of this report, the bearing was seized and damaged. Although the bearing was honed to a larger clearance prior to testing of both oils, doubt exists whether the spread of the eccentricity data may have been increased because of the poorer quality of the bearing bore.

Arguments Supporting the Experiments

The arguments supporting the experimentally observed behavior are as follows:

(1) The experimental data shown in figure 8 are among the first to be presented in an attempt to compare the load capacity of a non-Newtonian fluid with that of a Newtonian one. The trend indicating a marked increase in load capacity for the compounded fluid may be due to a physical phenomenon not yet fully understood or shown in figure 1, but which results from the complex rheological structure of the fluid. For example, figure 1 is related to a unidirectional shear rate, whereas the condition in a plain bearing involves varying components in the axial and rotative directions associated with a curved flow path within the film.

(2) A question arises as to whether it may be assumed in the analysis that at a given point in the film the fluid has different viscosity values in two perpendicular directions of flow because of different shear rates in the two directions. In either the capillary viscometer or the tapered-plug type the streamlines are parallel, and the measured viscosity is in the common direction of the streamlines. In the bearing film, however, the streamlines are curved so that the question arises as to whether the viscosity may be determined from components or should be determined from the shear rate of the velocity profile in the streamline direction. In the streamline direction the velocity profile is warped making a mathematical evaluation of shear rate from this profile difficult. It is not unreasonable to suppose that, in the non-Newtonian fluid with a polar long chain-molecular structure, the fluid possesses different directional properties including directional viscosity characteristics.[1]

(3) The assumptions used in the mathematical hydrodynamic analysis appear rational but certain of the assumptions lack rigor. The velocity profile in the axial direction is assumed parabolic as in a Newtonian fluid although it is known to be modified for a non-Newtonian fluid. Also,

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[1]A paper by Selby (ref. 6) and its discussions contain a number of references to this problem, including a reference to an unpublished early version of the present report which contained curves equivalent to figures 7 and 12.

the assumption of a linear velocity profile in the circumferential direction according to the short-bearing approximation may be too limiting an assumption to show the difference in the load capacity between the two types of fluid.

(4) Although the experimental apparatus for measuring eccentricity lacks the accuracy desirable for these tests and scatter in the measured eccentricities is to be expected, the scatter does not appear to be random; but, rather, it indicates a trend showing a superior performance for the non-Newtonian fluid. The two types of oil were tested in the same bearing with the same clearance and at the same journal speeds so that the W
data may be compared on the basis of only a change in type of oil. The 1
same bearing, reworked after damage, was used for both oils. 4
0

(5) In the tests of eccentricity a difference is possible in the film thicknesses of the two oils between the rotating journal surface and the contacting lever used to sense journal position. If there is such a difference, however, it is difficult to explain why the eccentricity data of figure 7 are so consistently equal for the two oils over the range of journal speeds of the experiments.

(6) Another possible source of error which could result in a bodily shift of the curves lies in the dependence of the eccentricity on the determination of the reference datum of measurement for each run. However, the probability is very small that this error would occur successively in the same direction for the number of datum determinations required.

Further Studies and Experimentation

Since the eccentricity behavior of the non-Newtonian oil is unexplained in this investigation, further experimentation and study should be made of the load-carrying capacity of such an oil. The experimental findings of this investigation should be checked by another method of experimentation using a more accurate method of measuring the film thickness. In recent years several experimenters have developed electrical methods for measuring minimum film thickness directly rather than by measuring eccentricity. Measurement of film thickness is a more sensitive method at high eccentricity ratios and would be especially appropriate for an investigation of the load capacity of the non-Newtonian fluid.

Although the most critical experimental device required is an accurate instrument to measure film thickness, it would also be ideal, if possible, to compare directly the load required for the two oils to produce the same film thickness with the same friction force. Tests of this kind would require a careful control of the viscosity of the normal oil by blending and/or by heat exchangers to control its temperature in order to achieve simultaneously a given friction force and a given film thickness.

Another suggestion less difficult to accomplish would be to use a system of valving which would permit a nearly instantaneous change of oil from the non-Newtonian to the Newtonian type in the same bearing without stopping. This would eliminate the reference datum error, if any, and permit direct comparison of the difference in the readings for the two oils. To prevent temperature changes in this method, blending of the Newtonian oil is suggested to give equal friction with either oil. With low oil flow some waste of oil can be tolerated in order to discard the mixture resulting during changes.

Although further investigation to check the experimental behavior in a plain bearing is desirable, equally important is a rheological study directed toward determining the special characteristics of the fluid which may account for its seemingly increased load-supporting capacity over that of a normal oil at the same friction force.

Considering the complexity of the streamline pattern in a journal bearing under load, testing the journal bearing itself seems at this time to be the only way to obtain this pattern; but some simpler methods to obtain each part of the condition separately would be useful. The tapered-plug viscometer is comparable to a concentric journal bearing without load or film pressure gradient. The capillary viscometer gives a modified parabolic profile fairly similar to one component of that in a bearing, but the smallest capillary holes used (0.010 inch) are of the order of 100 times the possible film thickness in a bearing. A viscometer having flow between flat plates separated by the thinnest possible distance would provide a practical method of obtaining small film thicknesses more like those in a journal bearing. The length of the flow path under a high pressure gradient in a bearing is only a fraction of the bearing axial length. Pressure gradients of 5,000 psi per inch in an alined bearing and 15,000 psi per inch in a misalined bearing are recorded in reference 7. The stationary flat plates would lack the rotational shear, but it might still be possible to obtain curved flow paths which may affect the two types of oil differently because of orientation effects of the long chain molecules (ref. 6).

CONCLUSIONS

The following conclusions may be drawn from the experimental and analytical results of this investigation of the effect of a non-Newtonian oil on friction and eccentricity ratio of a plain journal bearing:

1. The viscosity of a non-Newtonian oil for use in predicting the friction on the stationary element of a loaded plain bearing can be estimated from existing high-shear-rate viscometer data by using an analytically determined average shear rate which takes into account the eccentricity of the bearing.

2. The experimental and analytical methods used give conflicting indications of the effect of a non-Newtonian oil on load capacity of a plain bearing, neither being of the accuracy necessary to warrant a definite conclusion; but they seem to indicate the possibility of an improvement in load capacity relative to friction which warrants further investigation.

Cornell University,
Ithaca, N. Y., May 8, 1959.

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TABLE 1.- ANALYTICAL VALUES OF VISCOSITY μ_{e_f} BASED

ON EQUAL FRICTION FORCE

Eccentricity ratio, n	1,000 rpm, 131° F			8,000 rpm, 169° F		
	μ_{e_f}, reyn	μ_{e_f}, centipoises	θ_e, deg	μ_{e_f}, reyn	μ_{e_f}, centipoises	θ_e, deg
0.3	2.05×10^{-6}	14.12	100	0.995×10^{-6}	6.86	95
.5	2.00	13.80	111	.985	6.80	112
.7	1.92	13.22	123	.965	6.65	123
.9	1.75	12.07	139	.935	6.45	137

TABLE 2.- ANALYTICAL VALUES OF VISCOSITY μ_{e_L} BASED

ON EQUAL LOAD

Eccentricity ratio, n	1,000 rpm, 131° F			7,000 rpm, 158° F		
	μ_{e_L}, reyn	μ_{e_L}, centipoises	θ_e, deg	μ_{e_L}, reyn	μ_{e_L}, centipoises	θ_e, deg
0.3	2.32×10^{-6}	16.0	90	1.266×10^{-6}	8.74	90
.5	2.15	14.82	107	1.155	7.97	110
.7	1.86	12.81	127	1.07	7.37	125
.9	1.54	10.61	159	.92	6.35	157

TABLE 3.- COMPARISON OF VISCOSITY FOR FRICTION μ_{e_f} AND FOR LOAD μ_{e_L}

OBTAINED ANALYTICALLY AT LOW AND HIGH ROTATIVE SPEED

Eccentricity ratio, n	μ_{e_f}, reyns	μ_{e_L}, reyns	μ_{e_L}/μ_{e_f}	Increase in load capacity, percent
1,000 rpm, 131° F				
0.3	2.05×10^{-6}	2.32×10^{-6}	1.132	13.15
.5	2.00	2.147	1.074	7.35
.7	1.92	1.86	.969	-3.12
.9	1.75	1.54	.880	-12.00
8,000 rpm, 169° F				
0.3	0.995×10^{-6}	1.100×10^{-6}	1.106	10.55
.5	.985	1.010	1.025	2.54
.7	.965	.950	.985	-1.56
.9	.935	.880	.941	-5.87

TABLE 4.- COMPARISON OF ANALYTICAL VALUES OF μ_{e_f} AND μ_{e_L}

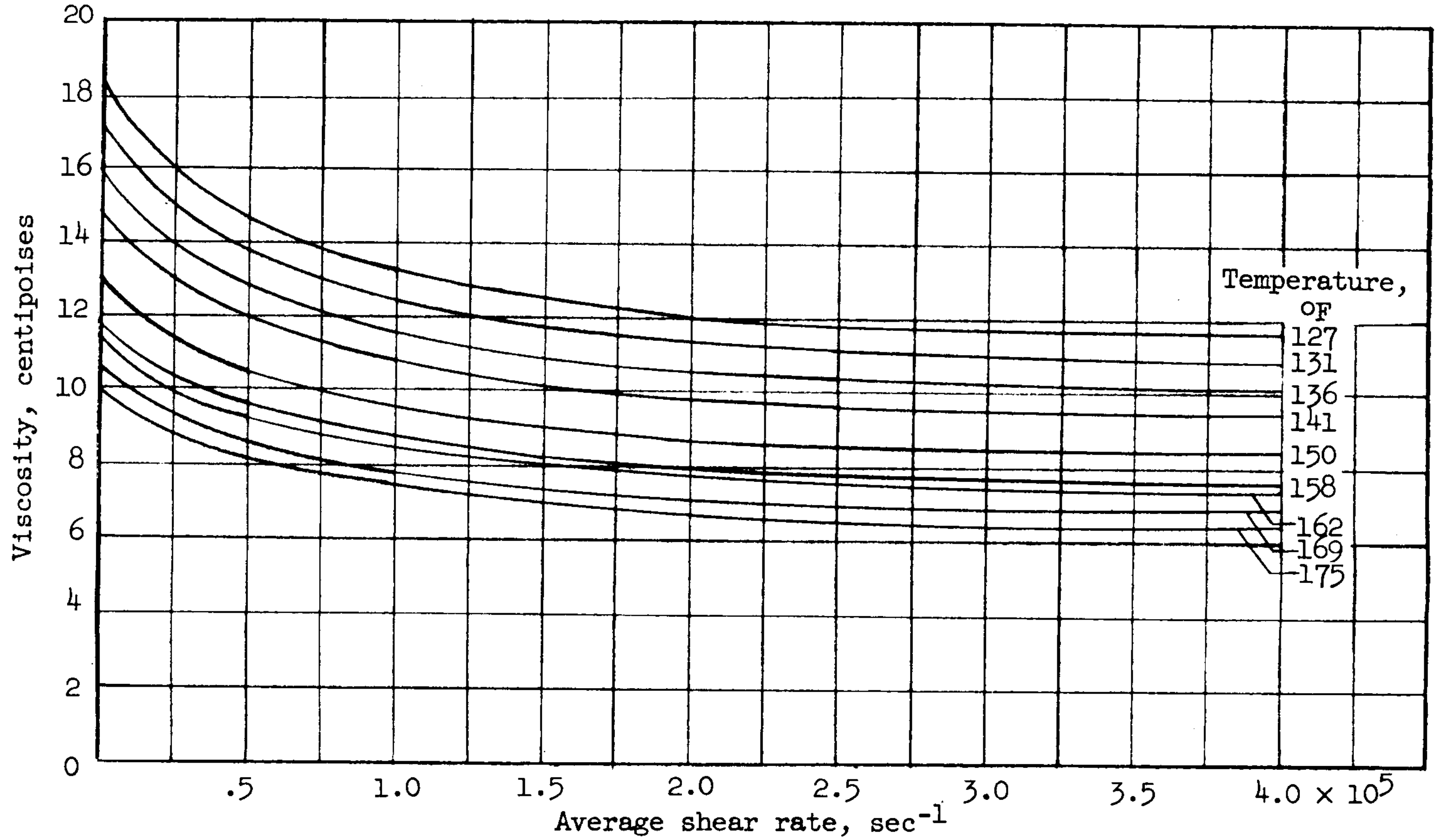
WITH VISCOSITY AT LOW AND HIGH SHEAR RATES

Journal speed, rpm	Average oil temperature, $^{\circ}F$	Range of experimental values		Zero-shear-rate viscosity, centipoises	High-shear-rate viscosity, centipoises
		μ_{e_f}, centipoises	μ_{e_L}, centipoises		
1,000	131	11.9 to 14.0	12.3 to 15.3	17.2	10.8
2,500	141	9.4 to 10.2	9.5 to 10.8	14.8	9.3
5,000	152	8.15 to 8.35	8.25 to 8.6	12.5	8.1
8,000	169	6.70 to 6.75	6.70 to 6.75	10.6	6.7

TABLE 5.- VISCOSITY DATA FOR TEST OILS

[Test oils and data were provided by Petroleum Refining Laboratory, Pennsylvania State University, through the courtesy of Dr. M. L. Fenske]

Finished composition property	Test temperature, °F	Value
ASTM 101 mineral oil; no polymer present		
Viscosity, centipoises	100	28.4
Viscosity, centipoises	210	4.30
Density, g/ml	100	0.857
Density, g/ml	210	0.820
ASTM chart slope	210 to 100	0.762
Viscosity index	----------	97
ASTM 104 mineral-oil-base blend; 5-weight-percent polymer		
Base stock viscosity, centipoises . .	100	7.5
Viscosity, centipoises	100	28.4
Viscosity, centipoises	210	7.46
Density, g/ml	100	0.854
Density, g/ml	210	0.816
ASTM chart slope	210 to 100	0.480
Viscosity index	----------	174



(a) Viscosity versus shear rate of non-Newtonian ASTM 104 oil, at constant temperature. Data from references 1 and 2 have been cross-plotted to temperatures obtained in current experiments.

Figure 1.- Viscosity variations.

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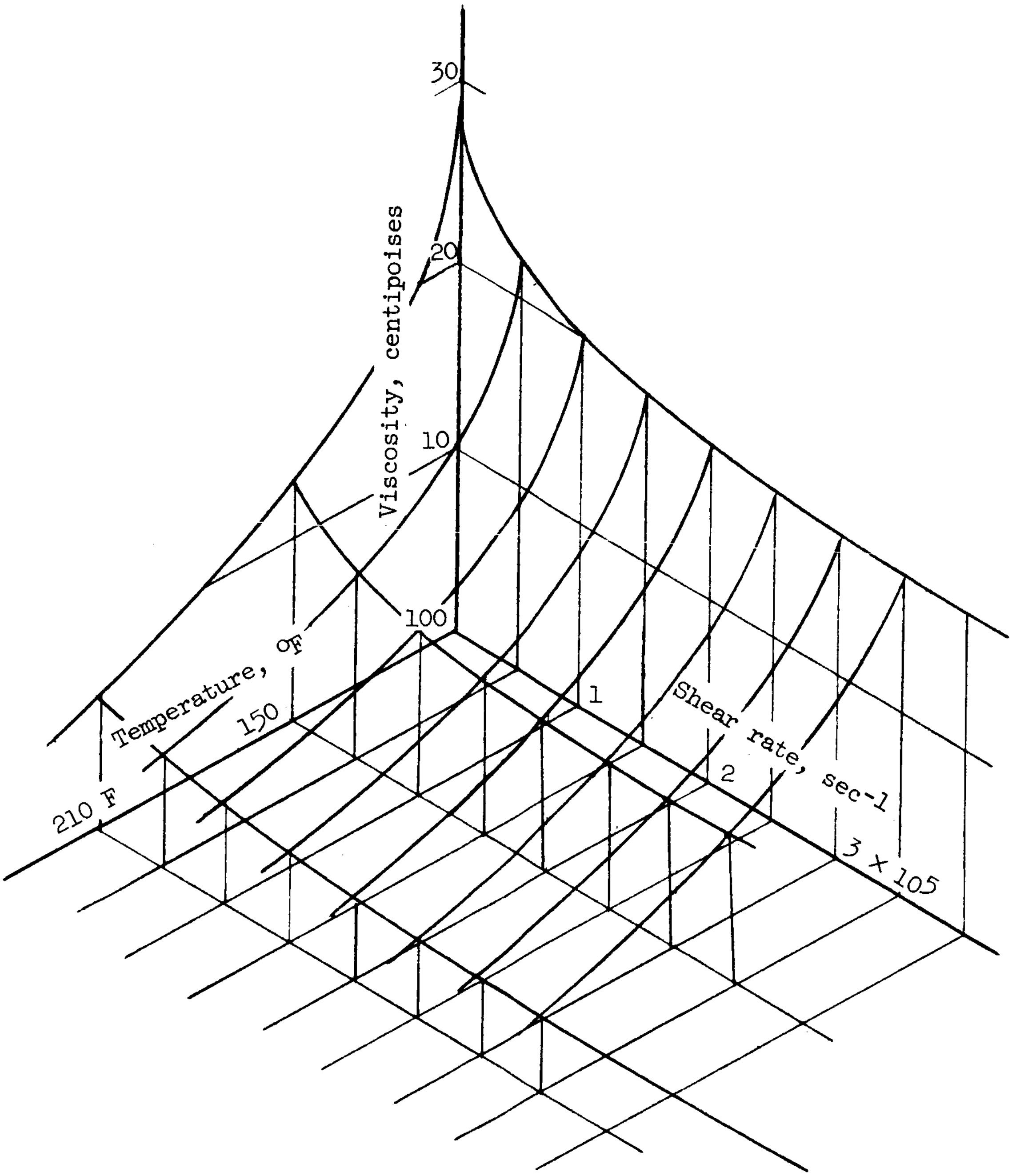
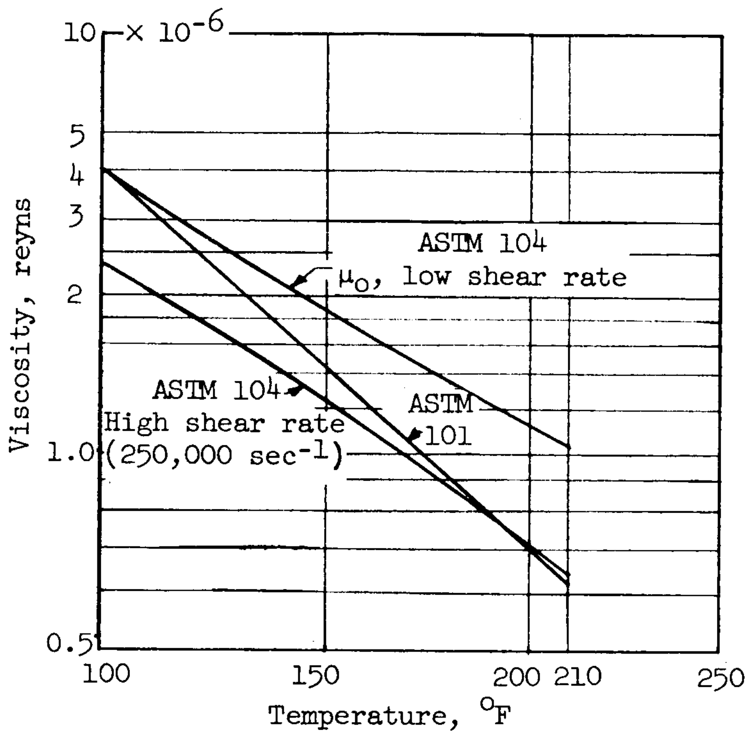


(b) Viscosity against temperature and shear rate for non-Newtonian ASTM 104 oil.

Figure 1.- Continued.



(c) Comparison of viscosity of Newtonian ASTM 101 oil with that of non-Newtonian ASTM 104 oil at low and high shear rates.

Figure 1.- Concluded.

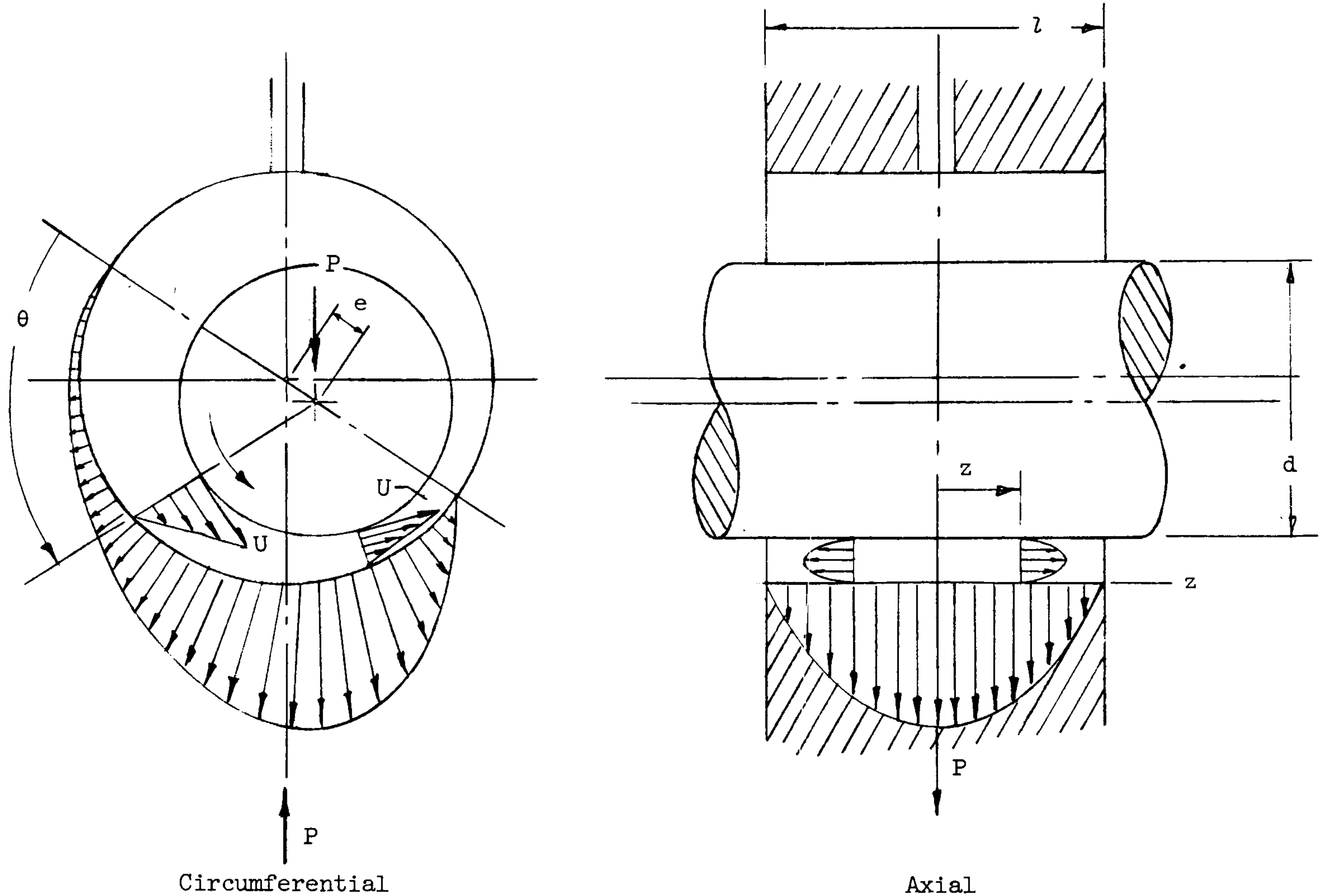


Figure 2.- Film-pressure distribution, velocity profiles, and variable shear rate in a loaded journal bearing showing analytical symbols and conditions based on short-bearing type of solution.

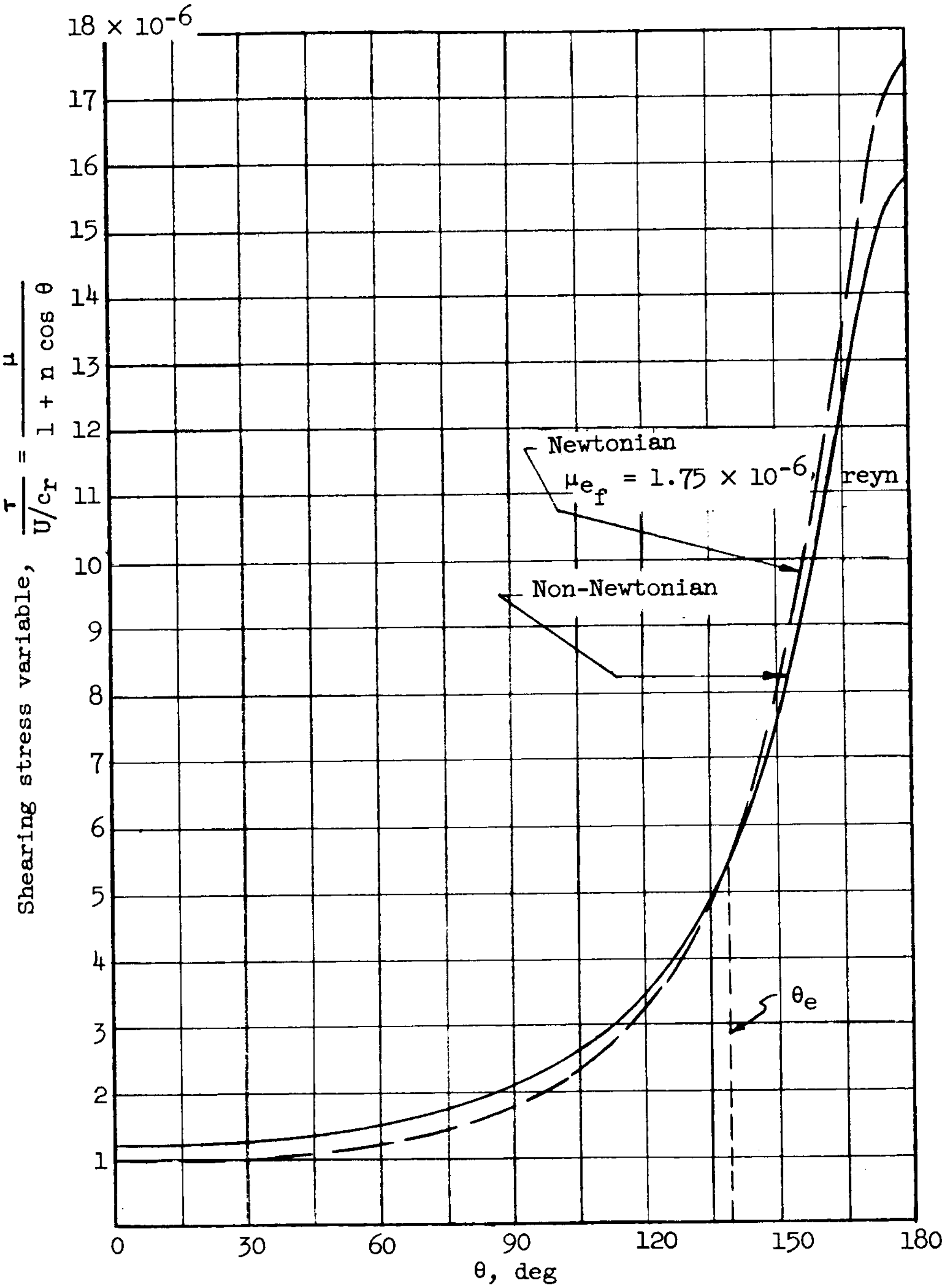


Figure 3.- Comparison of analytical shearing stress distribution for non-Newtonian and Newtonian oils in a bearing loaded to an eccentricity ratio $n = 0.90$ when friction forces are equal. Data for non-Newtonian ASTM 104 oil taken from figure 1 at 131° F; $N = 1{,}000$ rpm; $d/c_d = 393$; $l/d = 1.0$.

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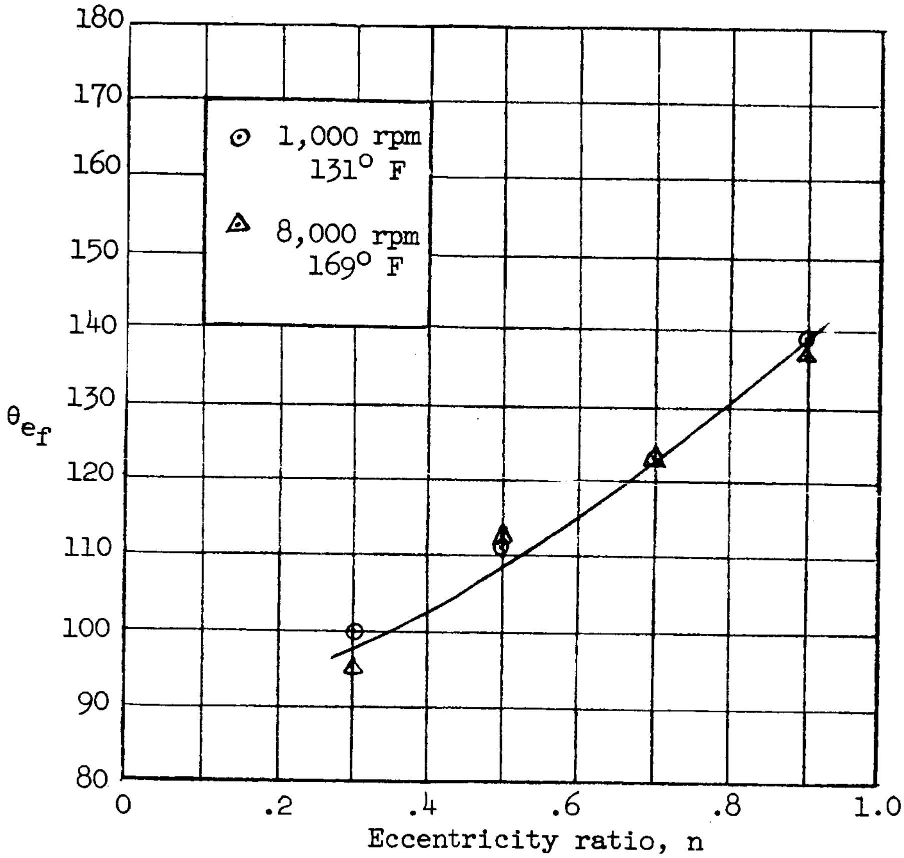


Figure 4.- Analytical values of angle θ_{e_f} at which equivalent viscosity μ_{e_f} occurs to give equal friction at two different speeds.

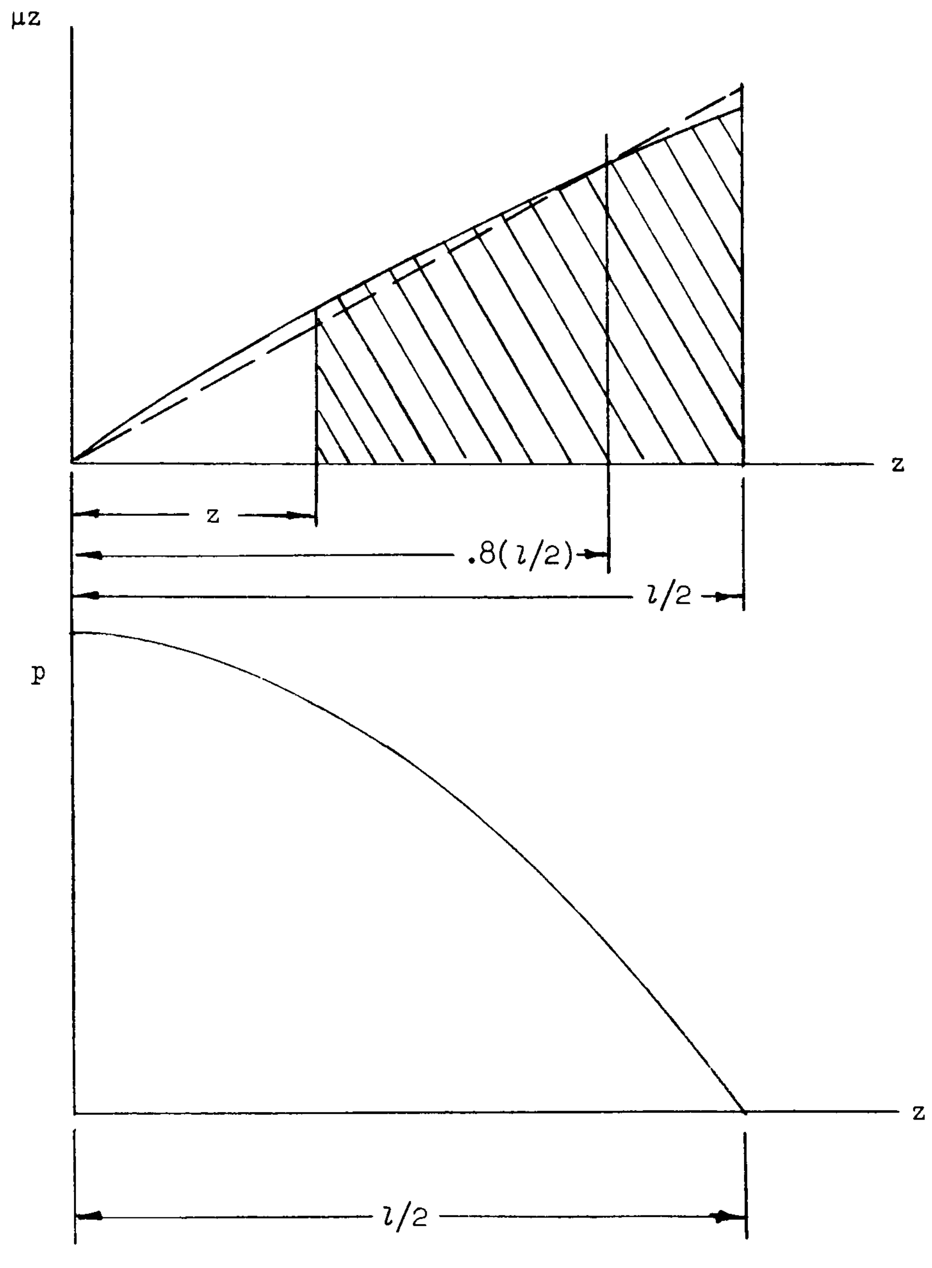


Figure 5.- Typical axial distribution of, above, the product μz and, below, film pressure obtained analytically for non-Newtonian oil.

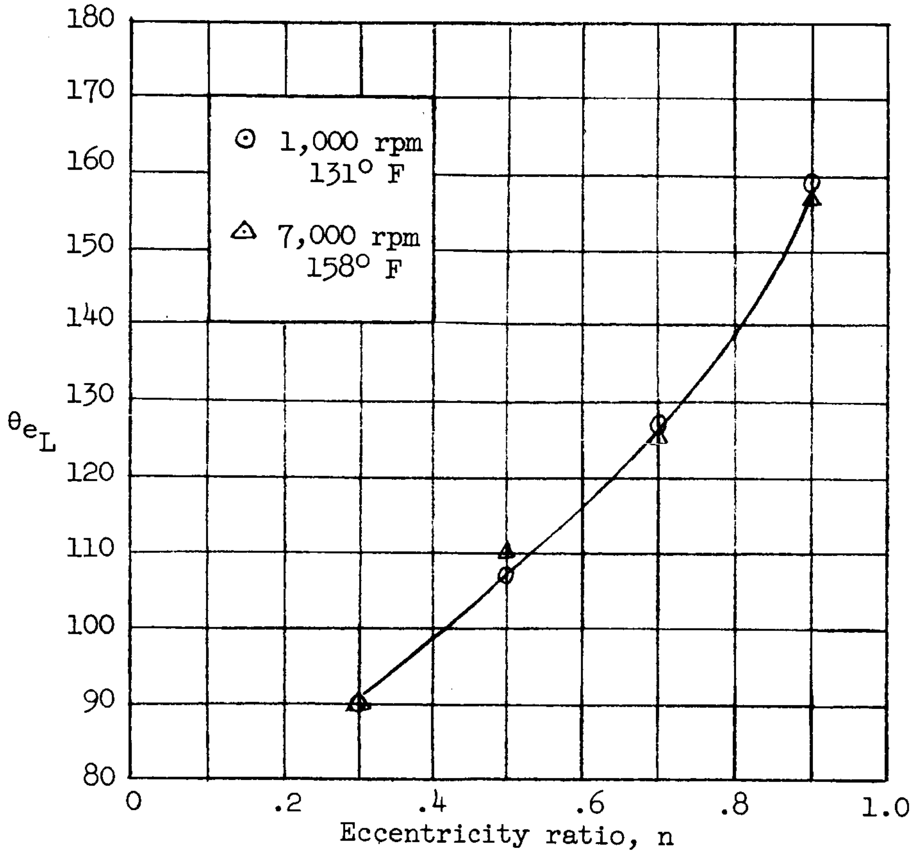
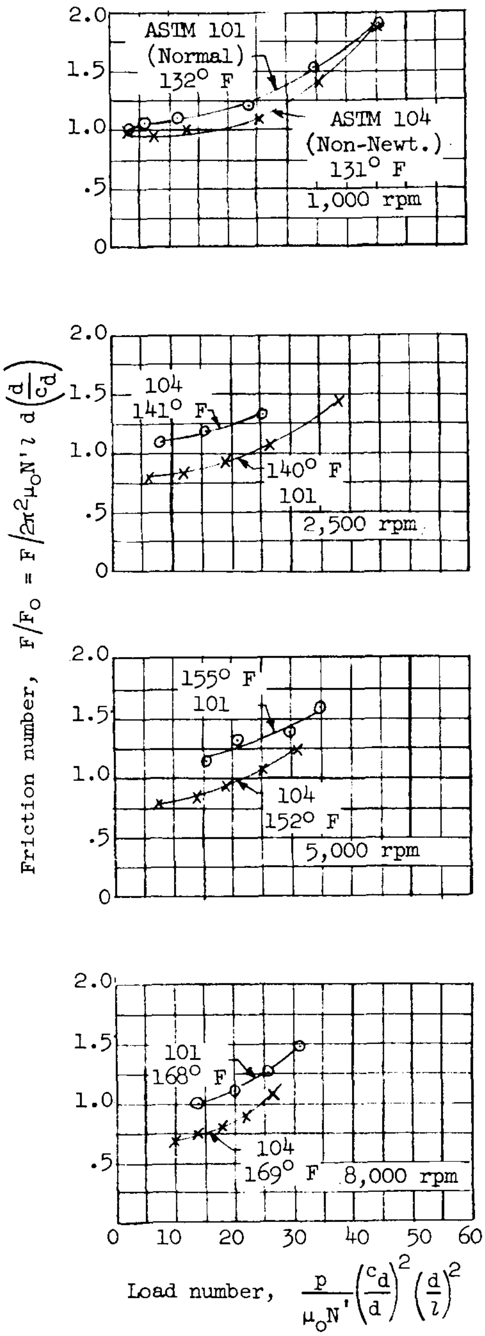
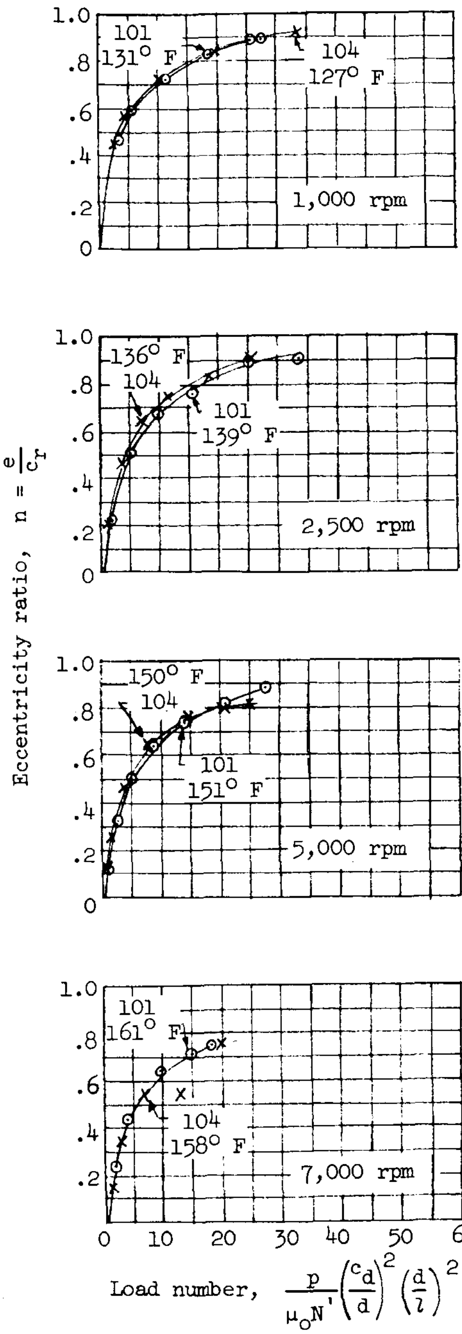


Figure 6.- Analytical values of angle θ_{e_L} at which equivalent viscosity μ_{e_L} occurs to give equal load at two different speeds.



(a) Friction data.



(b) Eccentricity-ratio data.

Figure 7.- Comparison of friction and eccentricity-ratio test data based on low-shear-rate viscosity (μ_o) of non-Newtonian oil, ASTM 104.

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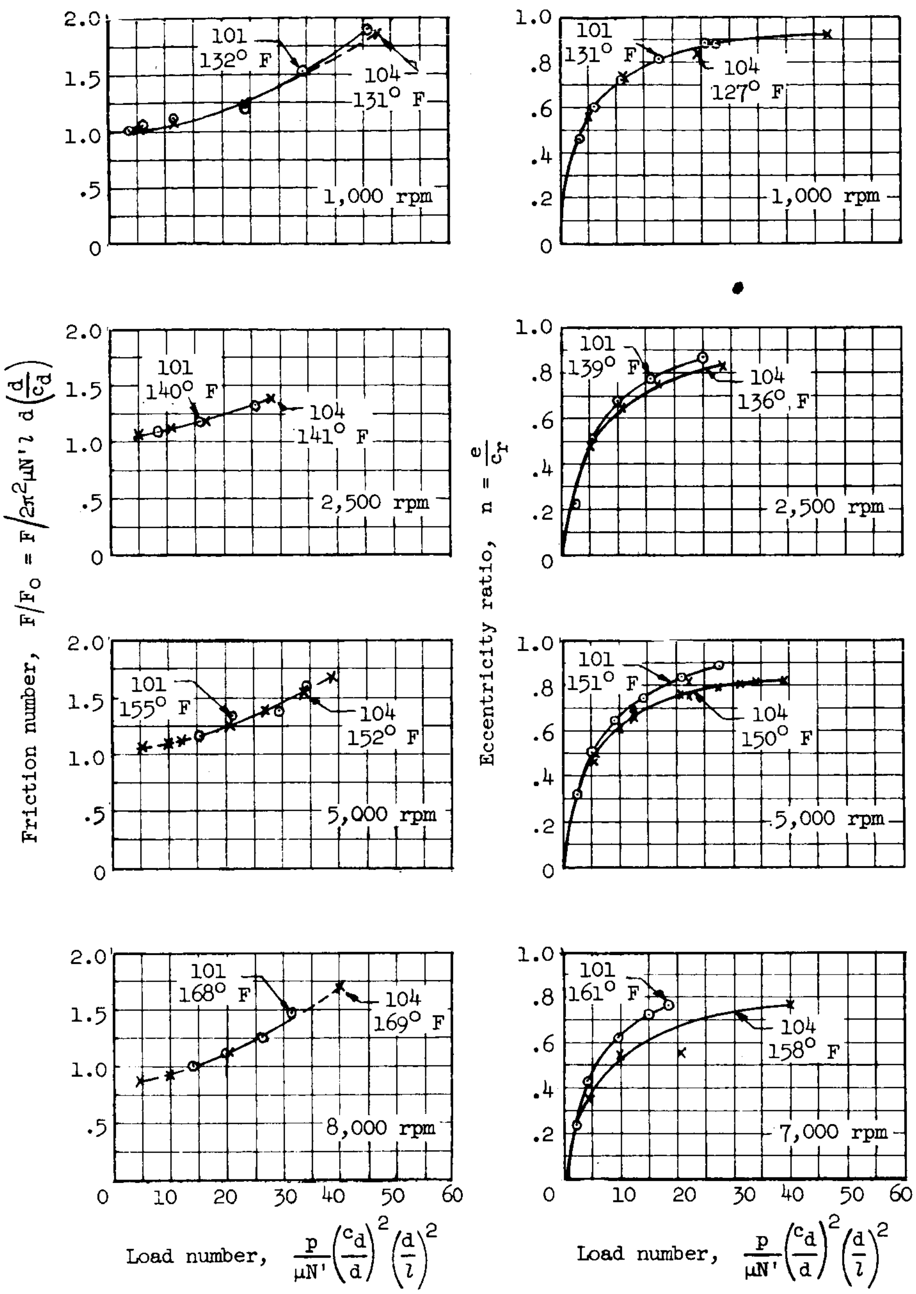
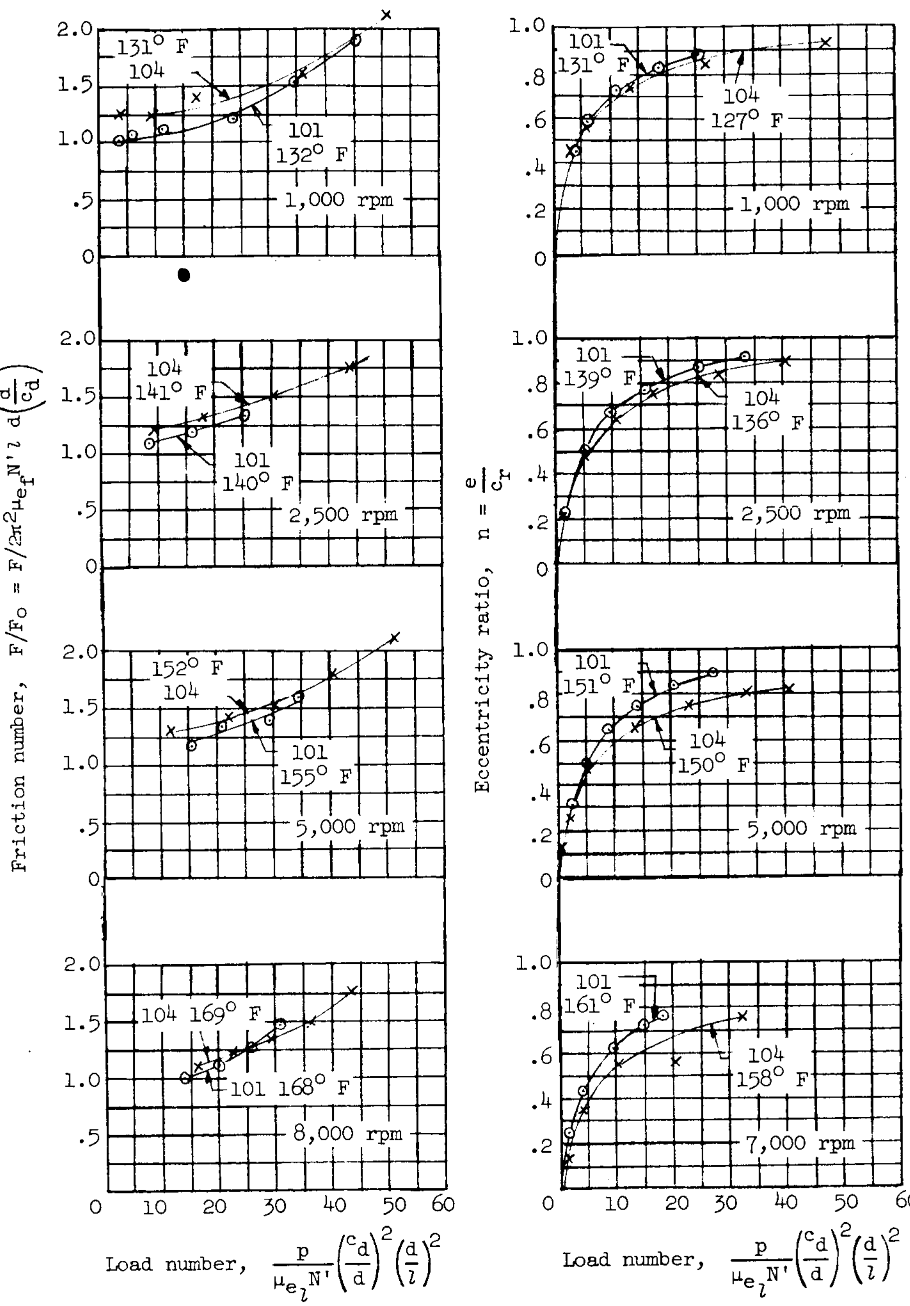


(a) Friction data. (b) Eccentricity-ratio data.

Figure 8.- Comparison of friction and eccentricity-ratio test data based on an experimentally equivalent viscosity for friction μ_{exp_f} for the non-Newtonian oil ASTM 104.



(a) Friction data.

(b) Eccentricity-ratio data.

Figure 9.- Comparison of friction and eccentricity-ratio test data based on analytically equivalent viscosities μ_{e_f} for friction and μ_{e_l} for load for non-Newtonian ASTM 104 oil.

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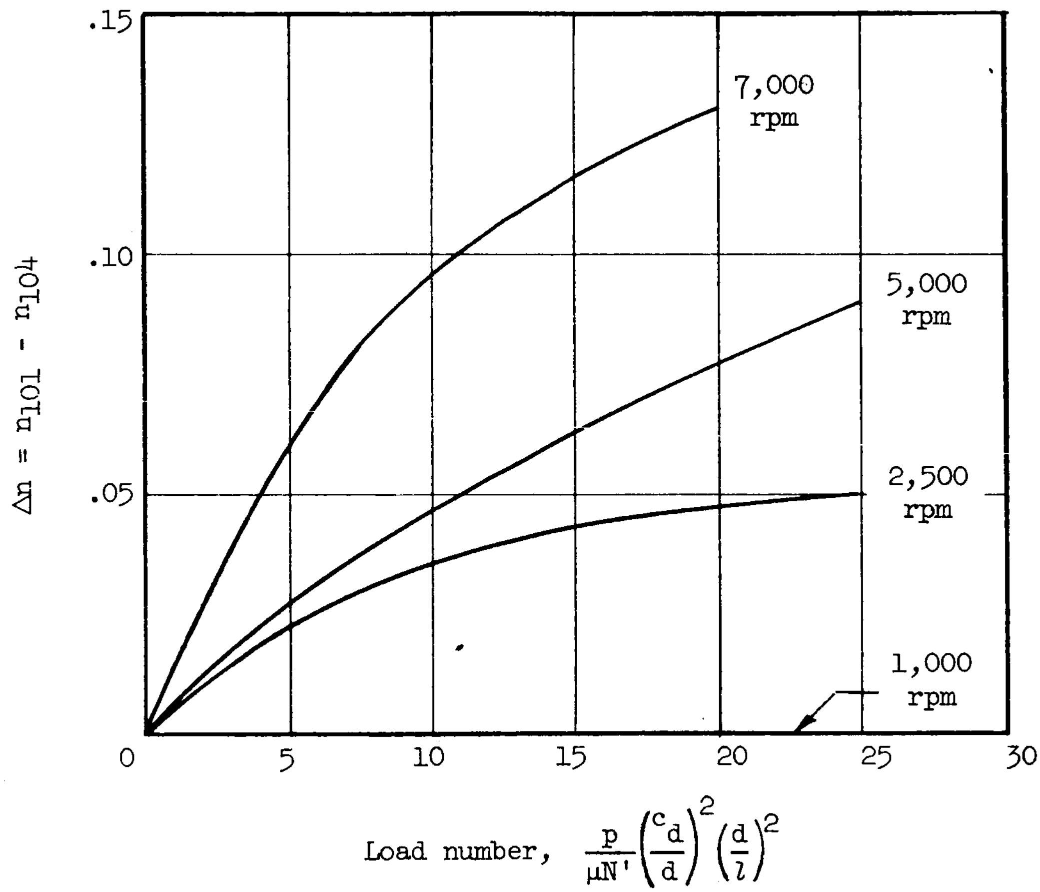

Figure 10.- Change of eccentricity ratio Δn at different speeds versus load number, comparing differences of two oils in figure 8.

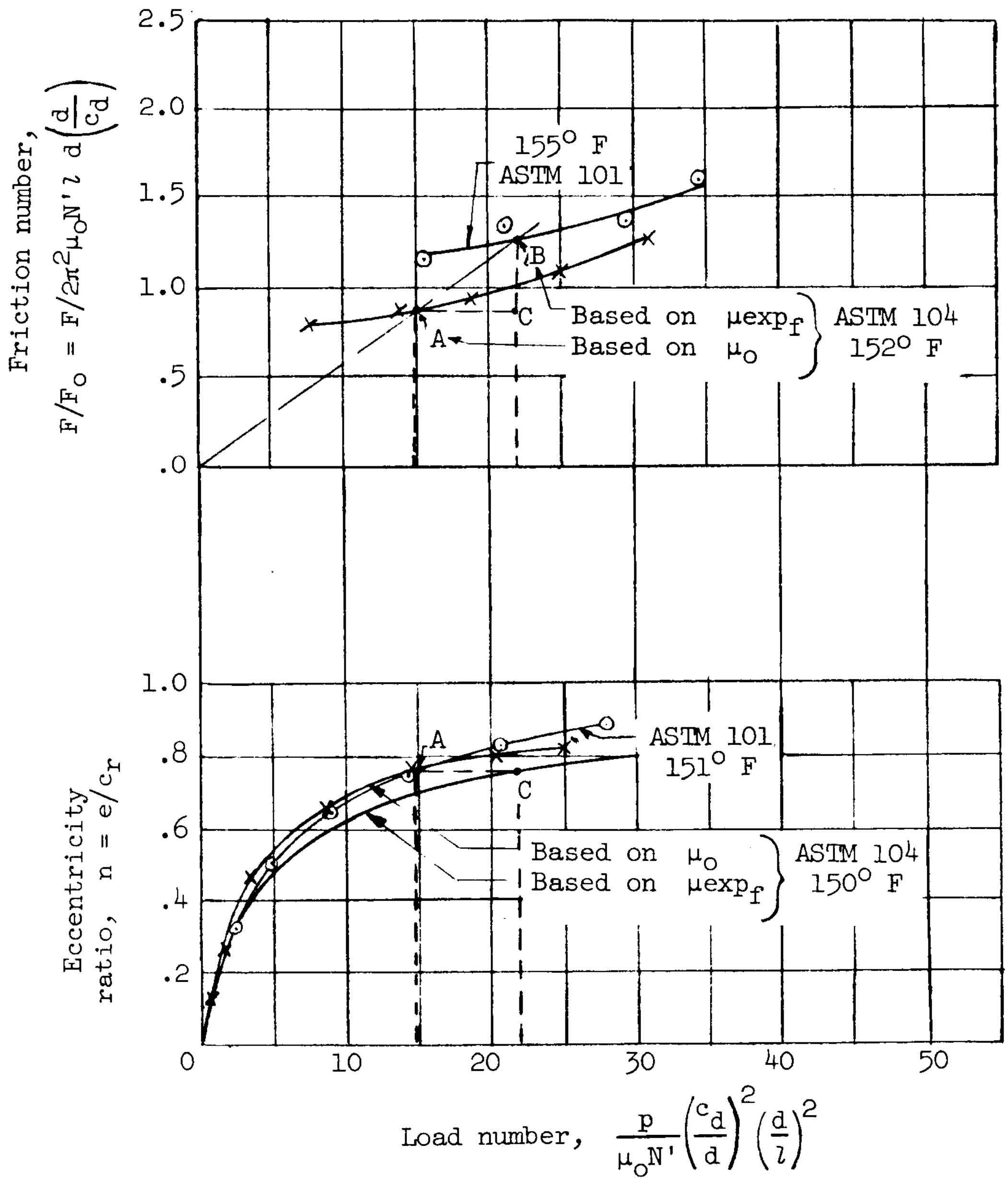

Figure 11.- Method of obtaining figure 8 graphically in order to base curves for non-Newtonian ASTM 104 oil on a viscosity μ_{exp_f} representative of friction data. Journal speed, 5,000 rpm.

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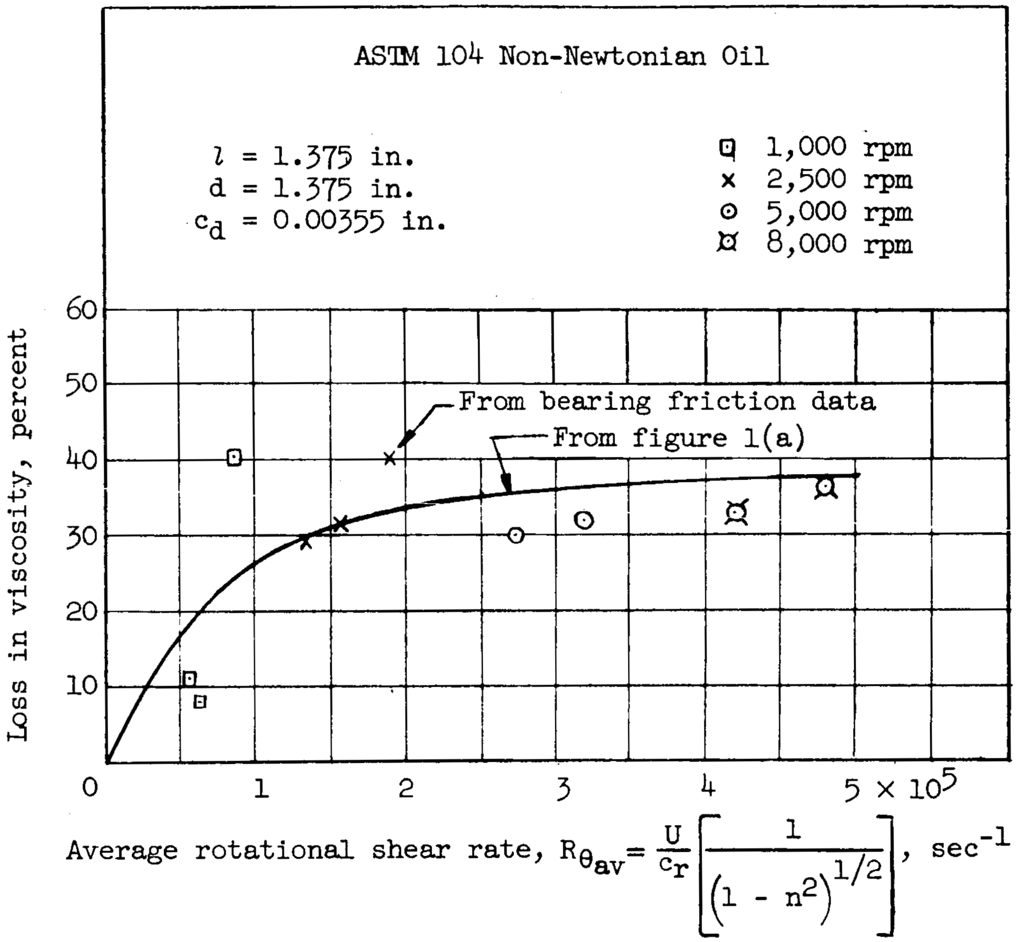

Figure 12.- Comparison of viscosity loss from bearing-friction data with estimated viscosity loss from high-shear-rate viscometer data in figure 1(a) based on average rotational shear rate corrected for eccentricity-ratio effect.